ISSN 1978-2497

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# Renkine-Cycle Analysis *( Estimate saving cost )* According to primary failures and Technical step to avoid more Deviations Ideal cycle of Re-heat the Steam generation (*according to case study of PLTU*)

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## ABSTRACT

*This paper discusses current applications of steam engines in industrial systems and new technologies which are improving their performance. The primary advantages of steam engine use come from applications where current technologies are either not appropriate or cannot be scaled down in size. Examples are given showing cases where it is more cost effective (and efficient) to use steam engines including dispatched power, use of a lower grade gas turbine with a steam engine bottoming cycle especially in analisis benefit cost and loses, every loses in supporting will effect amount cost loses, we will calculate some effect of failure with economic analisis, so we will taken advantages if we improve every point of failure, some failure will description with solutions and how Much money will saving This results in available steam power being failure. But how much our loses will calculated. And step ahead was how to improve eficiency to avoid lost energy, we shown real calculate to know how mauch we can save its by our step*

**Key word** : *combined cycle (CC), steam turbines (STs) gas turbines (GTs). waste heat recovery boilers (WHRB), heat recovery steam generators (HRSG), superheater & blowdown*

### A. General Benefits of Steam Engines

The discussion of steam engines begins necessarily by considering the importance of steam itself. Not quite a magic material, steam has the benefit of a large latent heat of vaporization. This is used in many ways, primarily using the change in phase for energy storage and energy release. As steam changes phase, it gives up energy without changing temperature –a phenomenon which is very useful in chemical processing, the development of power cycles and in heat exchanger design.Power systems utilizing steam also have the benefit of combustion external to the power prime mover. combustion. The obvious advantage of external combustion engines is enormousfuel flexibility. This principle appears in the design of nearly all new combined cycle power plants which can achieved thermal efficiencies greater than 50% without cogeneration (even higher with it!).

### B. Limitations of Steam Turbines

The question then turns to the prime mover. With high pressure, high temperature steam, one has really to choose between stream turbines or reciprocating or rotary engines.. Steam turbines are still ideal candidates for power, but there are tradeoffs from progressing up to turbine technology. The higher efficiency turbines have

- **Small turndown ratios** – turbines need to rotate at high speeds and are usually synchronized at 3600 rpm. With reduced pressure or flow rates, the fluid mechanics on the blading changes reducing system performance.

- **Slow startup times** – blading and heat exchangers associated with turbines are **very** thin,but held together with large supporting bolts. To reduce thermal shock, startup times arelong and the trend is toward even longer times on newer models.
- **Large capital costs and lingering economies of scale** – the complex nature of steam turbine systems, including significant water treatment, low pressure condensers, and high temperature and pressure tubing make initial costs high and reward larger installations where the ancillary costs are shared.
- **High cost for out-of-service time** – because of the large axis and horizontal orientation of **most** steam turbines, there is always the concern for warping of the shaft if the turbine is out of service for any reasonable length of time, resulting many cases in standby rotation of the shaft during its down period.

**C. Beneficial Aspects of Steam Engines**

Steam engines, in most cases, resolve these limitations and are the best chance for scale reduction in combined cycle power applications. In both DG and CHP applications, when steam is involved, turbines become questionable when the power output is below about 1 MW. A list of permance advantages for steam engines would include:

- **Fuel** flexibility - allows for a portfolio fuel approach. This includes thermal solar with very attractive installation costs since the rest of the system already exists. One must recognize that with many biomass and other waste fuels, there may be more emissionsconcerns with sulphur, mercury and other non-standard pollutants.
- **Sensitivity to** load - System efficiency is insensitive to load and can respond to rapidchanges in steam conditions
- **Low pressure combustion** – combined with no preheating allows for modest combustion temperatures and very little problem with NOx. Some new designs include coating the steam generator surfaces with an oxidation catalytic layer so that other emissions such as CO and unburned hydrocarbons can be reduced in a cost effective way
- **Modest speeds, pressures and temperatures -** are all in ranges which allow for safe and flexible operations Wear, noise, and maintenance are all improved because of low piston speeds.
- **Out-of-service and Startup** - startup is fast and steam engines can be out-of-service for long periods of time
- **Water Issues** - very modest water quality issues which results in reduced feedwater treatment costs. Also, the ability to handle wet steam in the pistons (even helping with lubrication) allows easy use of saturated steam

**D. principal steam Power Plant**

Much of the electricity is produced in steam power plants. spite efforts to develop alternative energy converters, electricity from steam willcontinue, for any years, to provide the power that energizes and world economies. We therefore begin the study of energy conversion systems with this important element of ndustrial society.Steam cycles used in electrical power plants and in the production of shaft powerin industry are based on the familiar Rankine cycle, studied briefly in most courses in thermodynamics we review the basic Rankine cycle and examine of the cycle that make modern power plants efficient and reliable.

**A Simple Rankine-Cycle Power Plant**

The most prominent physical feature of a modern steam power plant (other than itssmokestack) is the *steam generator*, or boiler, as seen in Figure.1. There the combustion, in air, of a fossil fuel such as oil, natural gas, or coal produces hot combustion gases that transfer heat to water passing through tubes in the steam generator. The heat transfer to the incoming water (*feedwater*) first increases its temperature until it becomes a saturated liquid, then evaporates it to form saturated vapor, and usually then further raises its temperature to create superheated steam. Steam power plants such as that shown in Figure.1, operate on sophisticated variants of the Rankine cycle.

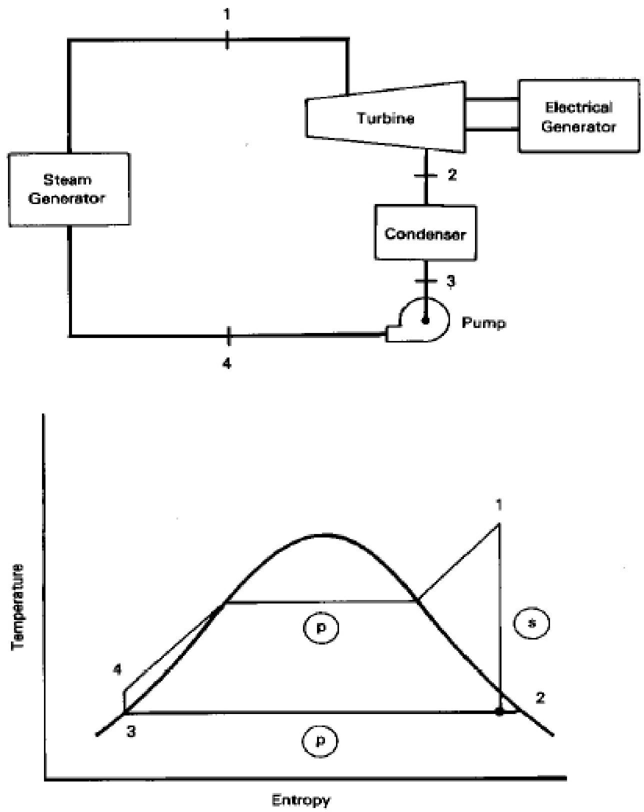


Figure1 Rankine cycle standar

In the simple Rankine cycle, steam flows to a turbine, where part of its energy is converted to mechanical energy that is transmitted by rotating shaft to drive an electrical generator. The reduced-energy steam flowing out of the turbine condenses to liuid water in the *condenser*. A feedwater pump returns the condensed liquid (*condensate*) to the steam generator We usually say that the working fluid (water) in the plant operates in a cycle or undergoes of cyclic process, as indicated in Figure1 In order to return the steam to the high-pressure of the steam generator to continue the cycle, the low- pressure steam leaving the turbine at state 2 is first condensed to a liquid at state 3 and then pressurized in a pump to state 4. The high pressure liquid water is then ready for its next pass through the steam generator to state 1 and around the Rankine cycle again.

**E. Rankine-Cycle Analysis**

In analyses of heat engine cycles it is usually assumed that the components of the engine are joined by conduits that allow transport of the working fluid from the exit of one component to the entrance of the next, with no intervening state change. It will be seen later that this simplification can be removed when necessary.It is also assumed that all flows of mass and energy are steady, so that the steady state conservation equations are applicable. This is appropriate to most situations because power plants usually operate at steady conditions for significant lengths of time. Thus, transients at startup and shutdown are special cases that will not be considered here.

Consider again the Rankine cycle shown in Figure 1. Control of the flow can be exercised by a throttle valve placed at the entrance to the turbine (state 1). Partial valve closure would reduce both the steam flow to the turbine and the resulting power output. We usually refer to the temperature and pressure at the entrance to the turbine as *throttle conditions*. In the ideal Rankine cycle shown, steam expands adiabatically and reversibly, or isentropically, through the turbine to a lower temperature and pressure at the condenser entrance. Applying the steady-flow form of the First Law of Thermodynamics for an isentropic turbine we obtain:

$q = 0 = h_2 . h_1 + w_t$ [Btu/lb$_m$ | kJ/kg]

where we neglect the usually small kinetic and potential energy differences between the
inlet and outlet. This equation shows that the turbine work per unit mass passing through the turbine is simply the difference between the entrance enthalpy and the lower exit enthalpy:

$w_t = h_1 . h_2$ [Btu/lb$_m$ | kJ/kg]

The power delivered by the turbine to an external load, such as an electrical generator,
is given by the following: Turbine Power = $m_s w_t = m_s(h_1 . h_2)$ [Btu/hr | kW]
where m$_s$ [lb$_m$ /hr | kg/s] is the mass flow of steam though the power plant. Applying the steady-flow First Law of Thermodynamics to the steam generator, we see that shaft work is zero and thus that the steam generator heat transfer is $q_a = h_1 . h_4$ [Btu/lb$_m$ | kJ/kg]

The condenser usually is a large shell-and-tube heat exchanger positioned below or adjacent to the turbine in order to directly receive the large flow rate of low-pressure turbine exit steam and convert it to liquid water. External cooling water is pumped through thousands of tubes in the condenser to transport the heat of condensation of the steam away from the plant. On leaving the condenser, the condensed liquid (called *condensate*) is at a low temperature and pressure compared with throttle conditions.

Continued removal of low-specific-volume liquid formed by condensation of the highspecific-volume steam may be thought of as creating and maintaining the low pressurein the condenser. The phase change in turn depends on the transfer of heat released to the external cooling water. Thus the rejection of heat to the surroundings by the cooling water is essential to maintaining the low pressure in the condenser. Applying
the steady-flow First Law of Thermodynamics to the condensing steam yields:

$q_c = h_3 . h_2$ [Btu/lb$_m$ | kJ/kg]

The condenser heat transfer $q_c$ is negative because $h_2 > h_3$. Thus, consistent with sign convention, $q_c$ represents an outflow of heat from the condensing steam. This heat is absorbed by the cooling water passing through the condenser tubes. The

condensercooling-water temperature rise and mass-flow rate $m_c$ are related to the rejected heat by: $m_s|q_c| = m_c c_{water}(T_{out} - T_{in})$ [Btu/hr | kW]
where $c_{water}$ is the heat capacity of the cooling water [Btu/lb$_m$-R | kJ/kg-K]. The condenser cooling water may be drawn from a river or a lake at the temperature $T_{in}$ and returned downstream at $T_{out}$, or it may be circulated through cooling towers where heat is rejected from the cooling water to the atmosphere. We can express the condenser heat transfer in terms of an overall heat transfer coefficient, $U$, the mean cooling water temperature, $T_m = (T_{out} + T_{in})/2$, and the condensing temperature $T_3$:
$m_s|q_c| = UA(T_3 - T_m)$ [Btu/hr | kJ/s]

It is seen for given heat rejection rate, the condenser size represented by the tube surface area $A$ depends inversely on (a) the temperature difference between the condensing steam and the cooling water, and (b) the overall heat-transfer coefficient. For a fixed average temperature difference between the two fluids on opposite sides of the condenser tube walls, the temperature of the available cooling water controls the condensing temperature and hence the pressure of the condensing steam. Therefore, the colder the cooling water, the lower the minimum temperature and pressure of the cycle and the higher the thermal efficiency of the cycle. A pump is a device that moves a liquid from a region of low pressure to one of high pressure. In the Rankine cycle the condenser condensate is raised to the pressure of the steam generator by *boiler feed pumps,* BFP. The high-pressure liquid water entering the steam generator is called *feedwater*. From the steady-flow First Law of Thermodynamics, the work and power required to drive the pump are:
$w_p = h_3 . h_4$ [Btu/lb$_m$ | kJ/kg] and Pump Power = $m_s w_p = m_s(h_3 . h_4)$ [Btu/hr | kW]
where the negative values resulting from the fact that $h_4 > h_3$ are in accordance with the thermodynamic sign convention, which indicates that work and power must be supplied
to operate the pump.

The net power delivered by the Rankine cycle is the difference between the turbine power and the magnitude of the pump power. One of the significant advantages of the Rankine cycle is that the pump power is usually quite small compared with the turbine power. This is indicated by the *work ratio*, $w_t / w_p$, which is large compared with one for Rankine cycle. ideal Rankine-cycle thermal efficiency in terms of cycle enthalpies as:
$\_{th} = (h_1 . h_2 + h_3 . h_4)/(h_1 . h_4)$ [dl]

In accordance with the Second Law of Thermodynamics, the Rankine cycle efficiency must be less than the efficiency of a Carnot engine operating between the same temperature extremes. As with the Carnot-cycle efficiency, Rankine-cycle efficiency improves when the average heat-addition temperature increases and the heatrejection temperature decreases. Thus cycle efficiency may be improved by increasing turbine inlet temperature and decreasing the condenser pressure (and thus the condenser temperature).

**F. Deviations from the Ideal .** *Component Efficiencies*

In a power plant analysis it is sometimes necessary to account for non-ideal effects such as fluid friction, turbulence, and flow separation in components otherwise assumed to be reversible..

**Turbine**

In the case of an *adiabatic turbine* with flow irreversibilities, the steady-flow First Law of Thermodynamics gives the same symbolic result as for the isentropic turbine in equation i.e., $w_t = h_1 . h_2$ [Btu/lb | kJ/kg] except that here $h_2$ represents the actual exit enthalpy and $w_t$ is the actual work of an adiabatic turbine where real effects such as flow separation, turbulence, irreversible internal heat transfers, and fluid friction exist., we seethat the turbine efficiency is:

$\_{turb} = (h_1 . h_2)/(h_1 . h_{2s})$ [dl] here $h_{2s}$, the isentropic turbine-exit enthalpy, is the enthalpy evaluated at the turbine inlet entropy and the exit pressure. For the special case of an isentropic turbine,

$h_2 = h_{2s}$ and the efficiency becomes 1.. $h_{2s}$, represents work lost due to irreversibility. Turbine isentropic efficiencies in the low 90% range are currently achievable in well-designed machines.

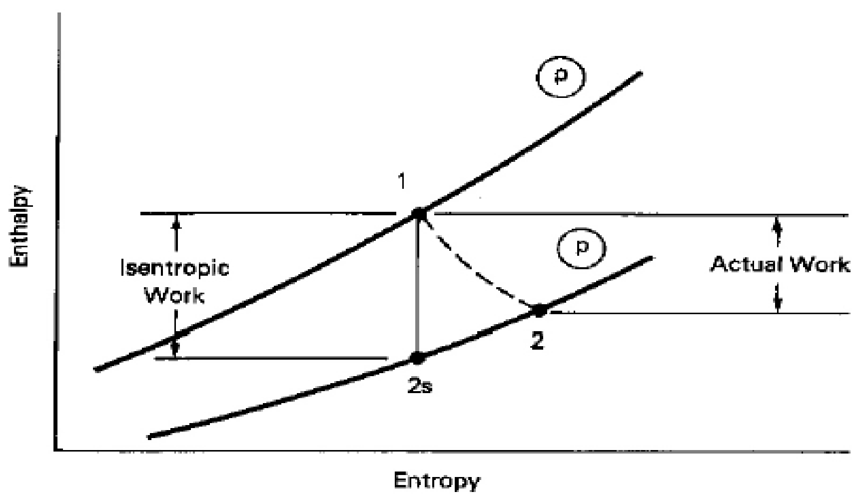


**Figure 2** Turbine isentropic efficiencies

Normally in solving problems involving turbines, the turbine efficiency is known from manufacturers. tests, and the inlet state and the exhaust pressure are specified. State 1 and $p_2$ determine the isentropic discharge state 2s using the steam tables. The actual turbine-exit enthalpy can then be calculated .Knowing both $p_2$ and $h_2$, we can then fully identify state 2 and account for real turbine behavior in any cycle analysis.

**Pump**

pump efficiency is the ratio of the isentropic work to theactual work input when operating between two given pressures. Applying Equation and the notation the isentropic pump ork, $w_{ps} = h_3 . h_{4s}$, and the *pump isentropic efficiency* is

$\_{pump} = w_{ps}/w_p = (h_{4s} . h_3)/(h_4 . h_3)$ [dl]

Note the progression of exit states that would occur in Figure 2.5 as pump efficiency increases for a fixed inlet state and exit pressure. It is seen that the pump lost work, given by $h_4 . h_{4s}$ decreases and that the actual discharge state approaches the isentropic discharge state.

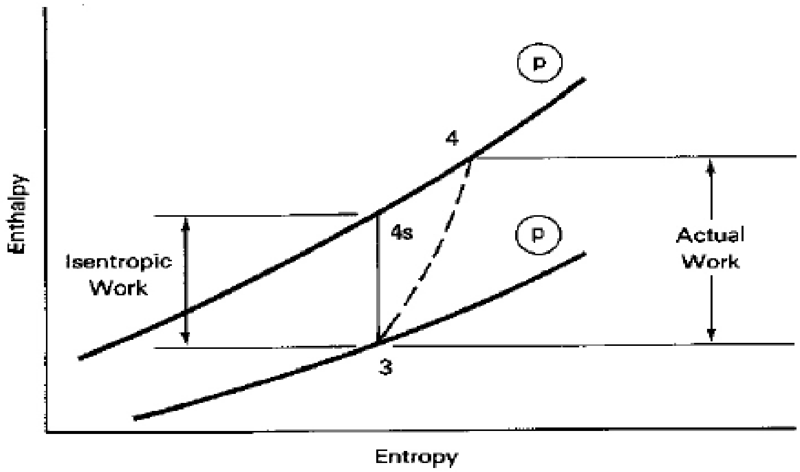


Figure 3 pump isentropic efficiencies

Using Equation and without consulting tables for subcooled water, we can then calculate the pump work from $w_p = v_3(p_3 . p_4)/\_p$ [ft-$lb_f$/$lb_m$ | kN-m/kg] if we Calculate the actual work and the isentropic and actual discharge enthalpies for an 80% efficient pump with an 80°F saturated-liquid inlet and an exit pressure of 3000 psia. We will solve that From the saturated-liquid tables, for 80°F, the pump inlet conditions are 0.5068 psia,48.037 Btu/$lb_m$, and 0.016072 $ft_3$/$lb_m$.Using Equation (2.9), we find that the pump work **is** $w_p$ = [0.016072(0.5068 . 3000)(144)]/0.8 = . 8677 ft-$lb_f$/ $lb_m$ or $w_p$ = . 8677/778 = . 11.15 Btu/$lb_m$. The actual discharge enthalpy is $h_4 = h_3 . w_p$ = 48.037 . (.11.15) = 59.19 Btu/$lb_m$ and the isentropic discharge enthalpy is $h_{4s} = h_3 . \_p\ w_p$ = 48.037 . (0.8)(. 11.15) = 56.96 Btu/$lb_m$. What is the turbine work, the net work, the work ratio, and the cycle thermal efficiency for the conditions if the turbine efficiency is 90% and the pump efficiency is 85%? What is the turbine exit quality? By the definition of isentropic efficiency, the turbine work is 90% of the isentropic turbine work = (0.9)(603.1) = 542.8 Btu/$lb_m$, the isentropic pump work is [(0.01614)(1 . 2000)(144)] / 778 = . 5.97 Btu/$lb_m$.

The actual pump work is then . 5.97/.85 = . 7.03 Btu/$lb_m$ and the work ratio is 542.8/| . 7.03| = 77.2, The cycle net work is $w_t + w_p$ = 542.8 . 7.03 = 535.8 Btu/$lb_m$.. Applying the steady-flow First Law of Thermodynamics to the pump, we get the enthalpy entering the steam generator to be $h_4 = h_3 . w_p$ = 69.73 . (. 7.03) = 76.76 Btu/$lb_m$.

The steam-generator heat addition is then reduced to 1474.1 . 76.76 = 1397.3 Btu/$lb_m$. and the cycle efficiency is 535.8/1397.3 = 0.383. Study of these examples shows that the sizable reduction in cycle efficiency from that in Example is largely due to the turbine inefficiency, not to the neglect of pump work.

**G Reheat and Reheat Cycles**

A common modification of the Rankine cycle in large power plants involves interrupting the steam expansion in the turbine to add more heat to the steam before completing the turbine expansion, a process known as *reheat*. steam from the high-pressure (HP) turbine is returned to the reheat section of the steam generator through the "cold reheat" line. There the steam passes through heated tubes which restore it to

ISSN 1978-2497



a temperature comparable to the throttle temperature of the high pressure turbine. The reenergized steam then is routed through the "hot reheat" line to a low-pressure turbine for completion of the expansion to the condenser pressure.

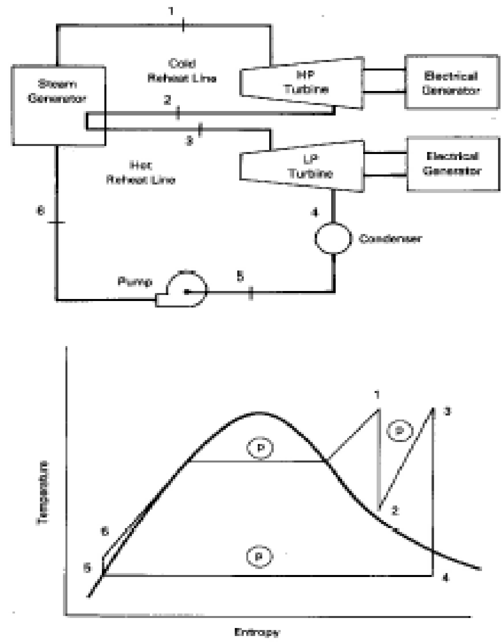


Figure 4 the reheat cycle

The cycle efficiency, the net work, and thermal efficiency of the reheat cycle is:

$$\_{th} = \frac{(h_1 \,.\, h_2) + (h_3 \,.\, h_4) + (h_5 \,.\, h_6)}{(h_1 \,.\, h_6) + (h_3 \,.\, h_2)} \quad [dl]$$

Reheat offers the ability to limit or eliminate moisture at the turbine exit. The presence of more than about 10% moisture in the turbine exhaust can cause erosion of blades near the turbine exit and reduce energy conversion efficiency. Study of Figure 2.6 shows that reheat shifts the turbine expansion process away from the two-phase region and toward the superheat region of the T-s diagram, thus drying the turbine exhaust. Reanalyze the cycle of Example (2000 psia/1000°F/1 psia) with reheat at 200 psia included. Determine the quality or degree of superheat at the exits of both turbines. Assume that reheat is to the HP turbine throttle temperature.we see that the properties of significant states are the following:

Table 1. Properties the steam tables

| State | Temperature (°F) | Pressure (psia) | Entropy (Btu/lb$_m$-°R) | Enthalpy (Btu/lb$_m$) |
|---|---|---|---|---|
| 1 | 1000.0 | 2000 | 1.5603 | 1474.1 |
| 2 | 400.0 | 200 | 1.5603 | 1210.0 |
| 3 | 1000.0 | 200 | 1.84 | 1527.0 |
| 4 | 101.74 | 1 | 1.84 | 1028.0 |
| 5 | 101.74 | 1 | 0.1326 | 69.73 |
| 6 | 101.74 | 2000 | 0.1326 | 69.73 |

Properties here are obtained from the steam tables and the Mollier chart as follows:

1. The enthalpy and entropy at state 1 are read from the superheated-steam tables at the given throttle temperature and pressure.
2. State 2 is evaluated from the Mollier diagram at the given reheat pressure and the same entropy as in state 1 for the isentropic turbine expansion.
3. 3. Reheat at constant pressure $p_3 = p_2$ to the assumed throttle temperature $T_3 = T_1$ gives $s_3$ and $h_3$. Normally, $T_3$ is assumed equal to $T_1$ unless otherwise specified.
4. The second turbine flow is also specified as isentropic with expansion at $s_4 = s_3$ to the known condenser pressure $p_4$.
5. The condenser exit (pump entrance) state is assumed to be a saturated liquid at the known condenser pressure.
6. Pump work is neglected here. The steady-flow First Law then implies that $h_6 = h_5$, which in turn implies the $T_6 = T_5$.

The turbine work is the sum of the work of both turbines:
(1474.1 - 1210) + (1527 - 1028) = 763.1 Btu/lb$_m$.
The heat added in the steam generator feedwater and reheat passes is
(1474.1 - 69.73) + (1527 - 1210) = 1721.4 Btu/lb$_m$.
The thermal efficiency then is 763.1/1721.4 = 0.443, or 44.3%.

## 6. Regeneration and Feedwater Heaters

The significant efficiency advantage of the Carnot cycle over the Rankine cycle is due to the fact that in the Carnot cycle all external heat addition is at a single hightemperature and all external heat rejection at a single low temperature. An internal transfer of heat that reduces or eliminates low-temperature additions of external heat to the working fluid is known as *regeneration.* A *feedwater heater* (FWH) is a heat exchanger in which the latent heat (and sometimes superheat) of small amounts of steam is used to increase thetemperature of liquid water (feedwater) flowing to the steam generator. This provides the internal transfer of heat mentioned above. An *open feedwater heater* is a FWH in which a small amount of steam mixes directly with the feedwater to raise its temperature. Steam drawn from a turbine for feedwater heating or other purposes is called *extraction steam*.

## 7. Cost-Effective Power Generation ( Estimated saving cost)

Modern Rankine-cycle power plants with 1,800 psig superheated steam boilers and condensing turbines exhausting at near-vacuum pressures can generate electricity with efficiencies of approximately 40%. Most steam users do not have the benefit of ultra-high-pressure boilers and cannot achieve such high levels of generation efficiency.

## A. Improve Boiler's Combustion Efficiency

Combustion efficiency is a measure of how effectively the heat content of a fuel is transferred into usable heat. The stack temperature and flue gas oxygen (or carbon dioxide) concentrations are primary indicators of combustion efficiency. The correct amount of excess air is determined from analyzing flue gas oxygen or carbon dioxide concentrations. Inadequate excess air results in unburned combustibles (fuel, soot, smoke, and carbon monoxide) while too much results in heat lost due to the increased flue gas flow—thus lowering the overall boiler fuel-to-steam efficiency.

Table 2. combustion efficiency

**Combustion Efficiency for Natural Gas**

| Excess % | | Combustion Efficiency | | | | |
|---|---|---|---|---|---|---|
| | | Flue gas temperature less combustion air temp, °F | | | | |
| Air | Oxygen | 200 | 300 | 400 | 500 | 600 |
| 9.5 | 2.0 | 85.4 | 83.1 | 80.8 | 78.4 | 76.0 |
| 15.0 | 3.0 | 85.2 | 82.8 | 80.4 | 77.9 | 75.4 |
| 28.1 | 5.0 | 84.7 | 82.1 | 79.5 | 76.7 | 74.0 |
| 44.9 | 7.0 | 84.1 | 81.2 | 78.2 | 75.2 | 72.1 |
| 81.6 | 10.0 | 82.8 | 79.3 | 75.6 | 71.9 | 68.2 |

Assumes complete combustion with no water vapor in the combustion air.

On well-designed natural gas-fired systems, an excess air level of 10% is attainable. An often stated rule of thumb is that boiler efficiency can be increased by 1% for each 15% reduction in excess air or 40°F reduction in stack gas temperature.A boiler operates for 8,000 hours per year and consumes 500,000 MMBtu of natural gas while producing 5,000 lb/hr of 150 psig steam. Stack gas measurements indicate an excess air level of 44.9% with a flue gas less combustion air temperature of 400°F. From the table, the boiler combustion efficiency is 78.2% (E1). Tuning the boiler reduces the excess air to 9.5% with a flue gas less combustion air temperature of 300°F. The boiler combustion efficiency increases to 83.1% (E2). Assuming a steam value of $4.50/MMBtu, the annual cost savings are:

Cost Savings = Fuel Consumption x (1 - E1/E2) x steam cost
= 29,482 MMBtu/yr x $4.50/MMBtu
= $132,671 annually

**B. Minimize Boiler Blowdown**

Minimizing your blowdown rate can substantially reduce energy losses, as the temperature of the blown-down liquid is the same as that of the steam generated in the boiler.. Excessive blowdown will waste energy, water, and chemicals. Blowdown rates typically range from 4% to 8% of boiler feedwater flow rate, but can be as high as 10% when makeup water has a high solids content.Assume that the installation of an automatic blowdown control system (see sidebar) reduces your blowdown rate from 8% to 6%. This example assumes a continuously operating natural-gas-fired, 150-psig, 100,000-pound-per-hour steam boiler. Assume a makeup

ISSN 1978-2497


water temperature of 60°F, boiler efficiency of 82%, with fuel valued at $3.00 per million Btu (MBtu), and the total water, sewage and treatment costs at $0.004 per gallon. Calculate the total annual cost savings. 100,000
Boiler Feedwater: Initial = = 108,695 lbs/hr (1 - 0.08) 100,000
Final = = 106,383 lbs/hr (1 - 0.06)
Makeup Water Savings = 108,695 – 106,383 = 2312 lbs/hr
Enthalpy of boiler water = 338.5 Btu/lb; for makeup water at 60°F = 28 Btu/lb
Thermal Energy Savings = 338.5 – 28 = 310.5 Btu/lb
Annual Fuel Savings = 2312 lbs/hr x 8760 hrs/yr x 310.5 Btu/lb x $3.00/MBtu = $23,007
0.82 x 106,Annual Water and Chemical Savings = 2312 lbs/hr x 8760 hrs/yr x$0.004/gal = $9,714,8.34 lbs/gal Annual Cost Savings = $23,007 + $9,714 = $32,721
The total annual energy savings are strongly dependent upon the facility energy cost and the hours per year of feedwater pump operation

Table 3 Annual energy saving

**Table 1. Annual Energy Savings when Using a Steam Turbine Feedwater Pump Drive[1]**

| Electricity Costs, $/kWh | Feedwater Pump Annual Operating Hours | | | | |
|---|---|---|---|---|---|
| | 2,000 | 4,000 | 6,000 | 7,000 | 8,760 |
| 0.03 | $4,105 | $8,210 | $12,310 | $14,365 | $17,975 |
| 0.05 | 13,525 | 27,050 | 40,570 | 47,330 | 59,230 |
| 0.075 | 25,305 | 50,605 | 75,910 | 88,560 | 110,830 |

[1] Savings are based upon operation of a 300-hp steam turbine drive with a steam rate of 26 lbs/hp-hr. A natural gas cost of $5.00/MMBtu is assumed.

**C. Insulate Steam Distribution and Condensate Return Lines**

Uninsulated steam distribution and condensate return lines are a constant source of wasted energy. The table shows typical heat loss from uninsulated steam distribution lines. Insulation can typically reduce energy losses by 90% and help ensure proper steam pressure at plant equipment. Any surface over 120°F should be insulated, Damaged or wet insulation should be repaired or immediately replaced to avoid compromising the insulating value. Eliminate sources of moisture prior to insulation replacement. Causes of wet insulation include leaking valves, external pipe leaks, tube leaks, or leaks from adjacent equipment. After steam lines are insulated, changes in heat flows can influence other parts of the steam system. In a plant where the value of steam is $4.50/MMBtu, a survey of the steam system identified 1,120 feet of bare 1-inch diameter steam line, and 175 feet of bare 2-inch line, both operating at 150 psig. An additional 250 feet of bare 4-inch diameter line operating,at 15 psig was found. From the table, the quantity of heat lost per year is:
1-inch line: 1,120 feet x 285 MMBtu/yr per 100 ft = 3,192 MMBtu/yr
2-inch line: 175 feet x 480 MMBtu/yr per 100 ft = 840 MMBtu/yr
4-inch line: 250 feet x 415 MMBtu/yr per 100 ft = 1,037 MMBtu/yr
Total Heat Loss = 5,069 MMBtu/yr
The annual operating cost savings from installing 90% efficient insulation is:
0.90 x $4.50/MMBtu x 5,069 MMBtu/yr = $20,530

### D. Return Condensate to the Boiler

When steam transfers its heat in a manufacturing process, heat exchanger, or heating coil, it reverts to a liquid phase called condensate. An attractive method of improving your power plant's energy efficiency is to increase the condensate return to the boiler. Returning hot condensate to the boiler makes sense for several reasons. As more condensate is returned, less make-up water is required, saving fuel, make-up water, and chemicals and treatment costs. Less condensate discharged into a sewer system reduces disposal costs. Return of high purity condensate also reduces energy losses due to boiler blowdown. Significant fuel savings occur as most returned condensate is relatively hot (130°F to 225°F), reducing the amount of cold make-up water (50°F to 60°F) that must be heated.A simple calculation indicates that energy in the condensate can be more than 10% of the total steam energy content of a typical system. The graph shows the heat remaining in the condensate at various condensate temperatures, for a steam system operating at 100 psig, with make-up water at 55°F.

### E. Annual Water, Sewage, and Chemicals Savings

Consider a steam system that returns an additional 10,000 lbs/hr of condensate at 180°F due to distribution modifications. Assume this system operates 8,000 hours annually with an average boiler efficiency of 82%, and make-up water temperature of 55°F. The water and sewage costs for the plant are $0.002/gal, and the water treatment cost is $0.002/gal. The fuel cost is $3.00 per Million Btu (MMBtu). Assuming a 12% flash steam loss*, calculate the overall annual savings.

= (1 – Flash Steam Fraction) x (Condensate Load in lbs/hr) x Annual Operating Hours x (Total Water Costs in $/gal) ÷ (Water Density in lbs/gal)

=(1 - 0.12) x 10,000 x 8,000 x $0.004

= $33,760

## 8. SUMMARY AND CONCLUSIONS

STs in CC applications have had to evolve significantly, from small 80 MW dual admission nonreheat configurations to multiple-pressure-admission reheat turbines with outputs reaching the 350 MW range. Because of this rapid evolution, many original small turbine designs have had to be modified or adjusted to compete in performance and capacity. Significant basic differences exist between STs designed for CCs and conventional Rankine cycle (RC) applications.

**First,** feedwater heaters are not normally used in the thermal design of the bottoming cycle for a CC; therefore, exhaust flows are much higher.

**Secondly,** the steam is added to the steam path flow at several pressures from the heat recovery steam generator (HRSG) to allow the maximum amount of heat to be extracted from the GT exhaust energy Regarding the failure of many STs to achieve their performance goals, Bechtel, as an experienced EPC contractor, has learned from this experience and has made provisions at the design stage as well as during execution to ensure that the total power plant performance guarantees are met. The integration of other major equipment (GT, HRSG, heat sink) was done to allow better-than-guaranteed performance.

**Thirdly**. **Estimated saving cost, there were some** analisis benefit cost and loses, every loses in supporting will effect amount cost loses, we will calculate some effect of failure with economic analisis, so we will taken advantages if we improve every point of failure, some failure will description with solutions and how Much money will saving This results in available steam power being failure. But how

much our loses will calculated. how to improve eficiency were : Improve Boiler's Combustion Efficiency, we cauld saving average $132,671 annually, . Minimize Boiler Blowdown Annual Cost Savings = $32,721 annually, Insulate Steam Distribution and Condensate Return Lines , The annual operating cost savings from installing 90% efficient insulation is:
0.90 x $4.50/MMBtu x 5,069 MMBtu/yr = $20,530, and last Annual Water, Sewage, and Chemicals Savings = = $33,760 of course if every point were improve their work and loses we would more save and increase eficincy **( more higher than our calculated above** !).

### References and Footnotes

1. "Recommended Rules for the Care and Operation of fluid", of the ASME Vessel Code, 1995.
2. "Recommended Guidelines for the Care of fluid dinamic", of the ASME Vessel Code, 1995.
3. Improving head loses sourcebook for industry Prepared by: Lawrence Berkeley National Laboratory Washington, DCResource Dynamics Corporation Vienna, VA
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